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MODELLING ADIABATIC CAPILLARY TUBES: A CRITICAL ANALYSIS

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ABSTRACT

A great variety of adiabatic capillary tube models are available in the open literature. The results of such numerical models are usually compared with experimental data and a good agreement is normally achieved. By using a specific computer model, called CAPILAR, it is shown that the common practice of selecting "traditional" values or expressions from the literature yields a considerable degree of freedom to such validation studies. It is strongly recommended that the upcoming studies focusing mainly in the stratospheric safe new refrigerants be conducted based on reliable sets of experimental data.

INTRODUCTION

An expansion device in common use in almost all small refrigeration systems is the capillary tube. This consists of a long hollow tube of drawn copper with an inside diameter between 0.5 to 2.0 mm and a length from 1 to 6 m.

Widespread use of capillary tubes did not begin until the introduction of the chlorofluorocarbon refrigerants. Previously, the principal refrigerant for small capacity applications was sulphur dioxide which required extremely small capillary tube inner diameters. These were subject to clogging by foreign matter and were not practical.

A capillary tube has no moving parts, nothing to wear out, and is simple and inexpensive. It also allows the pressures in the system to equalize during the off cycle, thus requiring compressor motors of low starting torque.

By the other hand, the capillary tube is not adjustable to changing load conditions and requires the refrigerant charge to be held within close limits. For this reason its use has been restricted to systems in which the load and refrigerant charge remain fairly constant, such as refrigerators and room air conditioners.

In spite of its strong influence on the refrigeration system performance most of the capillary tubes are still selected by a trial - and - error process. A longer tube than desired is first installed in the system. If it does not work properly it is shortened and another trial made. After several attempts at this cut - and - try process, one may be fortunate enough to accidentally find a workable capillary tube.

In an attempt to solve this problem a great variety of capillary tube computer models have been developed and validated against specific sets of experimental data. In doing so, empirical coefficients and correlations are usually selected from the literature. This practice, of course, yields a considerable degree of freedom to the validation exercise.

The purposes of this work are twofold: Firstly a computer model, called CAPILAR, will be introduced, and secondly the sensitivity of this program to the friction factor, underpressure of vaporization, etc will be assessed, in an attempt to identify areas requiring further experimental studies.

PRESSURE AND TEMPERATURE DISTRIBUTIONS IN ADIABATIC CAPILLARY TUBES

The capillary tube, although physically simple, is a very complex expansion device. This can be confirmed by the work of Shultz [1], where the relevant works on capillary tubes, performed until 1985, are presented.

A literature survey shows that the first works on capillary tubes were performed in the forties [2,3] and that until the sixties they were limited to the flow qualitative analysis, due mainly to the existing computing limitations.

The first major study of capillary tube flow was done by Bolstad and Jordan [4], in 1948. In this work, CFC-12 was circulated through various capillary tubes. The experimental apparatus was designed to allow the simultaneous control of the pressure and temperature of the refrigerant entering the capillary tube and also the evaporator pressure. Thermocouples were soldered to the outside at various positions along the tube. For one of the tubes studied, pressure measurements were also taken along the tube at two feet intervals.

With subcooled liquid entering the capillary tube, the observed pressure and temperature distributions were similar to that shown in Figure 1, where the saturation temperature scale corresponds to the pressure scale along the vertical axis.

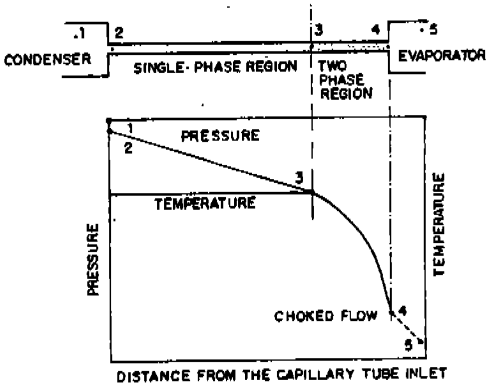

Fig. 1 - Pressure and temperature distribution along a capillary tube [4]

At the tube entrance there is a slight pressure drop, due mainly to the abrupt change in the cross sectional area, which was usually unreadable on the gages used by Bolstad and Jordan [4]. From point 2 to point 3 the pressure decreases linearly while the temperature is kept constant. Point 3 is the flash point, where the flow reaches saturated conditions. From this point up to the end of the capillary tube, the pressure drop per unit length increases as the end of the capillary tube is approached, and the temperature shows a correspondent drop. From point 4 to point 5, Bolstad and Jordan [4] observed the existence of a choked outlet condition and showed that after choking occurred, further lowering of the evaporator pressure had little effect on the mass flow rate. This situation, also called critical condition, corresponds to the point where the refrigerant entropy reaches its maximum value. The flow through the capillary tube cannot be further accelerated when the choked conditions are established. The pressure in this situation must be adequately calculated in order to correctly determine the mass flow rate through the capillary tube.

PROPOSED MODEL

The following assumptions were set up for the present model formulation: i) the capillary tube is a straight, horizontal constant inner diameter tube, ii) flow through the capillary tube is one-dimensional, homogeneous and adiabatic, iii) the refrigerant is free of oil, and iv) metastable flow phenomena are neglected.

Single Phase Flow Region

The length of the single-phase flow region of the capillary tube, L_{sf} can be determined by:

$$L_{sf} = \{[(p_1 - p_3)\,\rho/N] - 1 - K\}\, D/f_{sf} \tag{1}$$

where $N = G^2/2$ and $G = \dot{m}/A$

The symbols $p_1, p_3, \rho, K, D, f_{sf}, \dot{m}$ and A, correspond to inlet pressure, saturation pressure, density, entrance loss factor, inner diameter, single-phase friction factor, mass flow rate and tube cross-sectional area, respectively.

According to the procedure indicated by Collier[5] and considering that $A_2/A_1 \rightarrow 0$ and that the refrigerant at the capillary tube inlet is in a subcooled condition, the entrance loss factor K takes the value of 0.5.

The friction factor in the single-phase region, f_{sf}, is calculated by Blasius' equation, for smooth tubes:

$$f_{sf} = 0.31\, Re^{-0.25} \tag{2}$$

Two-Phase Flow Region

Applying the momentum conservation equation to a fluid element in the two-phase flow region, and integrating between points 3 and 4, yields:

$$L_{tf} = \left[2ln(\rho_4/\rho_3) - 2\int_3^4 \rho/G^2\, dp\right] D/\bar{f}_{tf} \tag{3}$$

The integral in equation (3) is numerically calculated by the Romberg's process. In order to do so the local density and the flow conditions at point 4 (choked or not) and, consequently, the local vapor quality are required.

The local vapor quality is determined by applying the energy equation to a fluid element. Thus,

$$x = \left\{ -h_{fg} - G^2 v_f\, v_{fg} + \left[(h_{fg} + G^2\, v_{fg} v_f)^2 - 2(G^2 v_{fg}^2)(h_f + G^2 v_f^2/2 - H_0)\right]^{0.5} \right\} / G^2 v_{fg}^2 \tag{4}$$

where h, v and H_0 are the enthalpy, specific volume and the stagnation enthalpy, respectively. The subscripts f and g, represent the liquid and vapor phases, respectively.

In order to determine whether the flow is choked, the local refrigerant entropy is calculated based on the fluid properties at saturation. Using a numerical procedure, the pressure, for which the entropy is maximum, can be easily determined. This choked pressure is then compared to the evaporating pressure (point 5), and the larger value will be used in equation (3) as p_4.

The average friction factor, in the two-phase flow region, is calculated by the procedure presented by Erth [6]. Erth selected 57 and 74 sets of data, from the work of Bolstad and Jordan [4] and Ungar et alii. [7] respectively. For each one of these data sets the pressure and temperature profiles were plotted, in a way similar to Figure 1. This procedure allowed the graphical determination of the liquid and two-phase regions lengths. From these data and using a regression analysis technique, the two-phase flow mean friction factor, $\bar{f}_{tf}$, was determined as a function of the inlet conditions only [6]:

$$\bar{f}_{tf} = 3.1\, Re^{-0.4}\, exp\left[(1 - x_i^{0.26})/2.4\right] \tag{5}$$

where

$$Re = G/\left[D(x_i\, \mu_{v,i} + (1 - x_i)\mu_{\ell,i}\right] \tag{6}$$

where μ is the refrigerant absolute viscosity and the subscript i, denotes inlet conditions. The thermodynamic and transport properties are taken from references [8-11].

The previous equations are solved only once when the objective is to determine the lengths L_{sf} and L_{tf}, from a known mass flow rate per unit of area, G. By the other hand, when the objective is to calculate the mass flow rate, through a specific capillary tube, the solving process has to be iterative though. The reason for such an iterative process is the dependence of the mass flow rate upon the friction factors and the dynamic pressure at the capillary tube inlet, which by their turn are dependent on the mass flow rate. The best iterative process, which yields the minimum CPU time, consists in firstly estimating the mass flow rate and then calculating the physical lengths L_{sf} and L_{tf}. The calculated total capillary length $(L_{sf} + L_{tf})$ is then compared with the design capillary length. If the physical capillary length is reached, the iterative process stops. Otherwise, the mass flow rate, initially guessed, is adjusted and the iterative process restarts.

COMPARISONS WITH EXPERIMENTAL DATA

The results, obtained with the CAPILAR program, will now be compared with the experimental data available in the literature. The major difficulty in doing so is the lack of information related to most of the experimental studies. Scott [12] made a great contribution in this respect, by organizing and analyzing most of the experimental data, published before 1976.

Figures 2, 3, 4 and 5 show a comparison between the computed results using the CAPILAR program and the measured data of Bolstad and Jordan [4], Whitesel [13], Mikol[14] and Ungar et alii. [7], respectively.

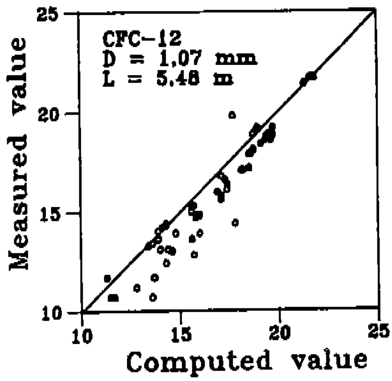

Fig.2 - Measured [4] versus computed mass flow rate in lbm/h

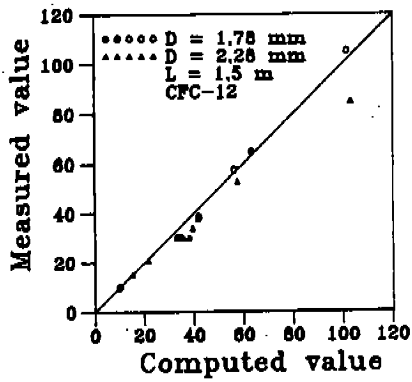

Fig.3 - Measured [13] versus computed mass flow rate in lbm/h

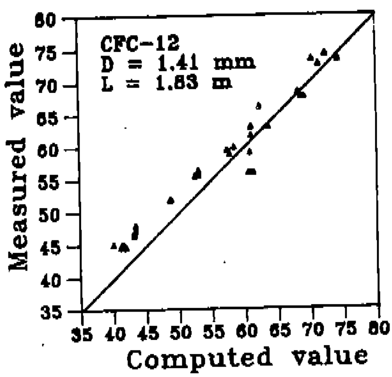

Fig.4 - Measured [14] versus computed mass flow rate in lbm/h

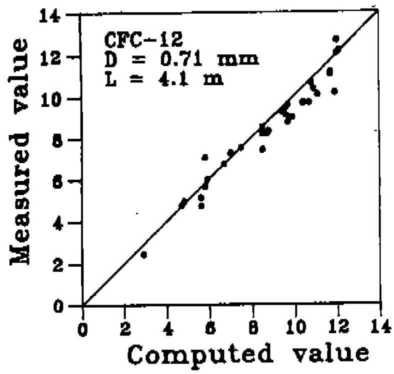

Fig.5 - Measured [7] versus computed mass flow rate in lbm/h

Distribution of errors is shown in histogramic fashion in Figure 6. A good measure of the model accuracy is that 120 runs or 82.8% of those calculated mass flow rates are within 10% of the experimental values.

Last December, Wijaya [15] published some experimental data using CFC-12 and HFC 134-a. To the authors' best knowledge this is the first published work presenting experimental data for HFC 134-a flow in capillary tubes. The comparison between the CAPILAR program and the experimental data, given by Wijaya [15], is presented in Tables I and II.

The evaporating pressures were not given in Tables I and II, since the experiments were made under choked flow conditions.

As one can see these comparisons are not so good as the ones shown in Figure 6.

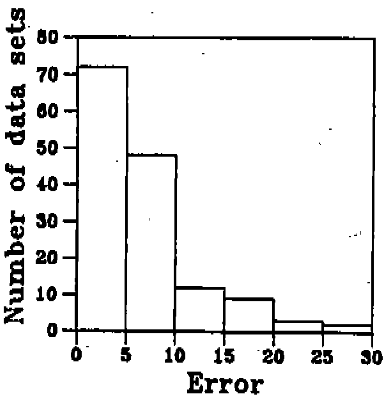

Fig.6 - Distribution of percent errors

Table I - Comparison with the experimental data of Wijaya [14] - CFC-12

Inlet Pressure (psia)	ΔT_{sub} ($^\circ F$)	$\dot{m}_{exp}$ (lbm/h)	$\dot{m}_{comp}$ (lbm/h)	Error %
131.6	9.6	11.05	8.73	-21.0
132.5	20.0	12.01	10.93	-9.0
132.8	30.1	13.16	12.57	-4.5
196.1	10.0	14.68	11.31	-22.9
197.6	20.4	15.89	13.69	-13.8
197.3	30.3	17.05	15.46	-9.3

Table II - Comparison with the experimental data of Wijaya [14] - HFC-134-a

Inlet Pressure (psia)	ΔT_{sub} ($^\circ F$)	$\dot{m}_{exp}$ (lbm/h)	$\dot{m}_{comp}$ (lbm/h)	Error %
140.14	10.4	11.53	8.83	-23.4
140.38	20.6	12.76	11.12	-12.8
139.84	30.1	13.67	12.59	-7.9
214.68	10.0	14.61	11.27	-22.8
214.96	20.3	16.27	13.71	-15.7
214.70	29.9	18.12	15.53	-14.3

SENSITIVITY ANALYSIS

The comparisons presented in the previous section are related to the original version of the CAPILAR program. Following the common procedure, adopted by many authors of other models, this work could be finished here, by concluding that the present model is good enough. Instead, a sensitivity analysis will be presented, including data and empirical correlations, in order to assess its impact on the program performance.

<u>Inner diameter of the capillary tube</u>

The model assumes that the cross sectional area of the capillary tube is circular, having a constant inner diameter along its entire length. This situation is rarely found in practice.

The ANSI/ASTM B-36/76 standard establishes a value of $\pm 25\mu m$ as the acceptable variation in the inner diameter, during the manufacturing process.

If such a variation is considered, in its extreme values for a capillary tube, operating with a condensing temperature of $54.4^\circ C$, an evaporating temperature of $-23.3^\circ C$ and a degree of subcooling of $5.5^\circ C$, the predicted mass flow rate displays the variation illustrated in Figure 7. Such a variation is even greater for shorter tube lengths and larger tube diameters.

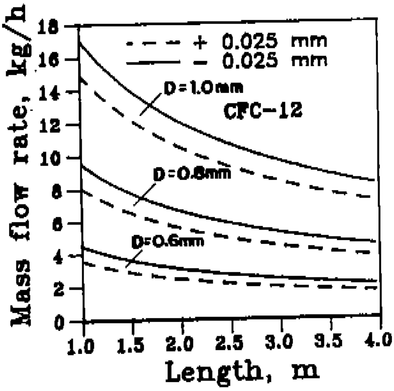

Fig.7 - Influence of the tube inner diameter on the mass flow rate

Therefore, in order to assure the desired accuracy, during the validation studies, it is essential to identify an "equivalent "inner diameter. This can be done by carefully filling up, with distilled water, and weighing the whole capillary tube [14], or from flow measurements with water and liquid CFC-11 [16].

Entrance loss factor

The entrance loss factor of 0.5, employed by the CAPILAR model, was derived from expressions published in the literature for an abrupt change in cross sectional area. However, most of the capillary tubes connections to the condenser tubes are different from the theoretical situation considered.

Scott [12] made several measurements to determine the contraction loss at the entrance of various capillary tubes, employing both hot and cold water. In spite of the considerable scatter data observed, a mean value of 0.4 was chosen.

Computational tests were made under the same conditions of Figure 7, but employing entrance loss factor ranging from 0.1 to 0.9. The results, thus obtained, were almost the same. This means that the absence of such information has little influence on the validation studies.

Metastable flow

The phenomenon of metastable flow, where the liquid exists at a temperature higher than the saturation temperature corresponding to its pressure, in capillary tubes, was firstly observed by Cooper et al. [17], in 1957 and later on by Mikol [14], in 1963.

During several years, this phenomenon was treated as an anomaly [6]. This situation changed with the works of Koizumi and Yokoyama [18], in 1980, and Kuijpers and Janssen [16], in 1983. Chen et alii. [19] and Li et alii. [20] published the most recent and complete work on metastable flow analysis [2], similar to the one presented by Kuijpers and Janssen [16], but also established a correlation for the determination of the pressure difference between the thermodynamic saturated point along the capillary tube and the point of inception of vaporization [19]. This pressure difference $(p_s - p_v)$ defined as the underpressure of vaporization, can then be calculated by:

$$\frac{(p_s - p_v)\sqrt{kT_s}}{\sigma^{3/2}} = 0.679\left(\frac{v_g}{v_g - v_f}\right)Re^{0.914}\left(\frac{\Delta T_{sub}}{T_{crit}}\right)^{-0.208}\left(\frac{D}{D'}\right)^{-3.18} \tag{7}$$

where

$$D' = \sqrt{\frac{kT_s}{\sigma}} * 10^4 \tag{8}$$

The symbols $k, T_s, T_{crit}, \Delta T_{sub}$ and σ represent the Boltzman's constant, absolute saturation temperature, critical temperature, degree of subcooling and surface tension, respectively.

Equations (7) and (8) were included in the CAPILAR model, by considering the flow as being in thermodynamic equilibrium conditions until point b (liquid) and from point b to c (vapor), (see Figure 8). This simplification lies on the fact that the regions II and III (see Figure 8) are really non-equilibrium regions.

The program was run, under the same conditions of Figure 7, but accounting for the effects of metastable flow. It can be noticed (see Figure 9) that, as expected, the mass flow rate through the capillary tube is increased, when the effects of the metastable flow are considered.

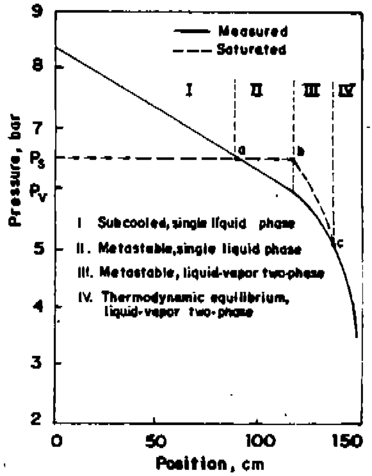

Fig.8 - Pressure profiles in capillary tubes, including the effects of metastable flow [19]

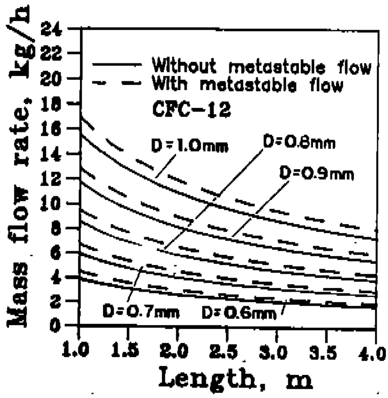

Fig.9 - Influence of the metastable flow on the mass flow rate

Simulations were also performed, using the data of Table I, but considering the metastable flow effects. In this case, the difference between numerical and experimental results decreases by only 3%.

It should be stressed that Chen et alii. [19] employed only two capillary tubes, both 1.5m long and 0.66mm and 1.17mm in diameter, respectively, and that the relative error involved in equation (7) is 26%.

Accounting for the metastable flow effects via computer models is essential for proper mass flow rate predictions, but extensive experimental studies are still required in order to improve the understanding of refrigerant flow through capillary tubes, before a quantitative analysis of the metastable flow effects can be generally accepted.

Single-phase friction factor

Mikol [14], studying the flow of CFC-12 through capillary tubes, concluded that this kind of expansion device cannot be considered smooth for purposes of friction factor calculation. In his experiments Mikol [14] used capillary tubes with a roughness of 0.000533mm, determined by two independent means for surface profilometer measurement.

Mikol [14] also concluded that Moody's diagram, for both laminar and turbulent flow, is applicable to single-phase flow through capillary tubes. The mathematical representation of Moody's diagram is the well known Colebook equation

$$f_{sf} = \left\{ \frac{1}{1.14 + 2\log\frac{D}{\varepsilon} - 2\log\left[1 + \frac{9.3}{Re\,(\varepsilon/D)\sqrt{f_{sf}}}\right]} \right\}^2 \qquad (9)$$

This equation is implicit in f_{sf}, so a iterative process is required.

Lin et alii. [21] observed that their friction factors were 20% higher than the values calculated by Blasius' equation. The absolute ronghness of their two capillary tubes were 0.002mm and 0.0035mm for the 0.66mm and 1.10mm ID capillary tubes , respectively. It should be stressed that such values are almost ten times greater than the values published by other authors [14, 6, 12].

The proposed correlation [21] takes the following form:

$$f_{s,f} = 8\left[\left(\frac{8}{Re}\right)^{12} + \frac{1}{(A+B)^{3/2}}\right]^{\frac{1}{12}} \tag{10}$$

where

$$A = \left\{2.457\,ln\left[\frac{1}{(7/Re)^{0.9} + 0.27\,E/D}\right]\right\}^{16} \tag{11}$$

and

$$B = \left[\frac{3750}{Re}\right]^{16} \tag{12}$$

Figure 10 shows the friction factor calculated according to Blasius, Colebrook and Lin et alii. [21] equations, as a function of the Reynolds number.

It can be observed that both Lin et alii. [21] and Colebrook's equation yield friction factors higher than the ones calculated by Blasius' equation. However, the mass flow rate predictions of the CAPILAR model are almost not affected by the use of any one of these correlations.

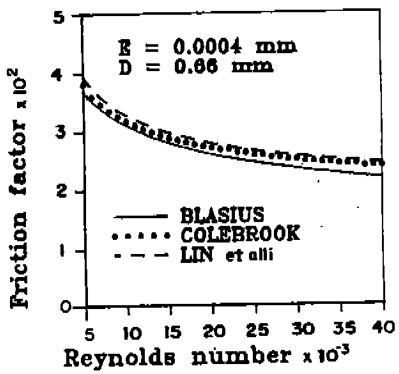

Fig.10 - Single-phase friction factors as a function of the Reynolds number.

Two-phase flow friction factor

The first works on capillary tubes modelling, employed the Blasius' equation, also in the two-phase region. In those works the absolute viscosity, appearing in the Reynolds number, was calculated as a function of the vapor quality, by different equations. Scott [12] presents those equations, as well as a comparison between the resultant friction factors.

In 1970, Erth [6] published his results (see equations (5) and (6)). The average relative roughness of the capillary tubes tested by Bolstad and Jordan [4] and by Ungar et alii. [7], were 0.00073 and 0.00034, respectively. As previously mentioned, those works [4, 7] formed the experimental basis of Erth's work.

Erth's equation has the advantage of being a function of the inlet conditions only. The mean friction factor, thus calculated, simplifies the numerical procedure (see equation (3)).

Lin et alii. [21] also presented an expression for the two-phase flow friction factor. This equation takes the following form:

$$f_{tf} = \Phi^2 f_{sf} \tag{13}$$

where f_{sf} is calculated according to equation (10)-(12) and Φ^2 is given by:

$$\Phi^2 = \left[\frac{\left(\frac{8}{Re_{tf}}\right)^{12} + \frac{1}{(A_{tf}+B_{tf})^{3/2}}}{\left(\frac{8}{Re_{sf}}\right)^{12} + \frac{1}{(A_{sf}+B_{sf})^{3/2}}} \right]^{\frac{1}{12}} \left[1 + x\left(\frac{v_g}{v_f} - 1\right)\right] \tag{14}$$

The parameters A and B are calculated by equations (11) and (12), respectively. The absolute viscosity appearing in the two-phase Reynolds number, R_{tf}, is calculated by equation (15).

$$\mu_{tf} = \frac{\mu_f \, \mu_g}{\mu_g + x^{1.4}\,(\mu_f - \mu_g)} \tag{15}$$

In order to compare the CAPILAR model predictions, using either the expression recommended by Erth [6], or the one presented by Lin et alii. [21], some alterations in the program were required. This is so, because Lin et alii. [21] equation yields the two-phase friction factor as a function of the local vapor quality, and the model employs an averaged value.

The pressure interval was then divided in 100 points. For each one of these points the vapor quality and the two-phase flow friction factor were determined, and an averaged value then determined.

The variation in the mass flow rate through a capillary tube, under the same conditions of Figure 7, when Erth's equation is replaced by the one presented by Lin et alii. [21], is shown in Figure 11.

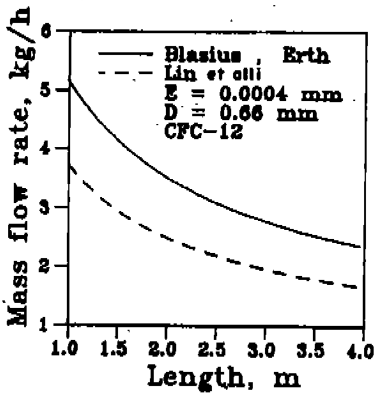

Fig.11 - Influence of the two-phase friction factor correlations on the mass flow rate

As one can see, the equation employed to evaluate the two-phase friction factor has a remarkable effect on the mass flow rate.

As a final comment, it should be mentioned that both capillary tube models, proposed by Erth [6] and by Chen et alii. [19], displayed a reasonable agreement with their own experimental data, in spite of the fact that the two-phase friction factors calculated by Lin et alii. [21], under the same conditions, are almost 10 times greater than the ones predicted by Erth [6].

CONCLUSIONS

A mathematical model of refrigerant flow through capillary tubes was presented.

The agreement between the model predictions and the experimental data, available in the literature, are reasonably good. Such a comparison is, however, dependent upon the equations chosen for building up the model.

The experimental data must always report the 'equivalent 'inner diameter and also the tube roughness, in order to enable an adequate validation exercise.

The two-phase friction factor in capillary tubes requires further experimental studies, since the available correlations display a large variation between their predictions.

The metastable flow through capillary tubes also requires deeper experimental investigation. The accuracy of the proposed correlation [19], for other tube diameters, lengths, operating conditions, tube roughness and refrigerant type, must be identified.

The experimental uncertainty related to the available data sets puts all the existing numerical models in the same level of prediction capacity. All the effort should be concentrated on generating reliable experimental information with adequate control of all the variables involved and also getting full understanding of the non-equilibrium phenomena of viscous flow through capillary tubes. After then there will be enough room for more refined numerical models.

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